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EXPERIMENTAL STUDY ON HEAT TRANSFER AND PRESSURE DROP IN PLATE HEAT EXCHANGER USING WATER-WATER

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ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ ТЕПЛООБМЕНА И ПЕРЕПАДА ДАВЛЕНИЯ В ПЛАСТИНЧАТОМ ТЕПЛООБМЕННИКЕ С ИСПОЛЬЗОВАНИЕМ СРЕД ТЕПЛОНОСИТЕЛЕЙ ВОДА-ВОДА

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Abstract. It would be misguiding to consider only cost in design of a heat exchanger because high maintenance costs increase total cost during heat exchanger's services life. Therefore, exergy analysis and energy saving are very important parameters in heat exchanger design. In this study, the effects of pate heat exchanger (PHE, Ridan HHN no. 04) on heat transfer, friction factor and exergy loss were investigated experimentally. Experiments were conducted from laminar flow condition to turbulence flow condition under counter flow condition. Reynolds number and Prandtl number were in the range of 220 <Re <2400 and 3 <Pr <7, respectively. Heat transfer, friction factor, and exergy loss correlations were obtained according to the experimental results.

Аннотация. Было бы ошибкой рассматривать только стоимость конструкции теплообменника, поскольку высокие эксплуатационные расходы увеличивают общую стоимость в течение срока службы теплообменника. Поэтому анализ эксергии и энергосбережения являются критическими параметрами конструкции теплообменника. В этом исследовании экспериментально изучено влияние пластинчатого теплообменника (PHE, Ridan HHN №04) на теплопередачу, коэффициент трения и потери эксергии. Эксперименты проводились от состояния ламинарного течения до состояния потока турбулентности в условиях встречного течения. Число Рейнольдса и число Прандтля находились в диапазоне 220 <Re <2400 и 3 <Pr <7, соответственно. Согласно результатам эксперимента, получены теплопередача, коэффициент трения и соотношение потерь эксергии.

Keywords: plate heat exchanger, heat transfer, convective heat transfer coefficient, exergy loss.

Ключевые слова: пластинчатый теплообменник, теплообмен, коэффициент конвективной теплопередачи, потеря эксергии.

Introduction

Heat exchanger is an important industrial device used to transfer heat between hot and cold stream for energy conservation. Types of heat exchanger used for energy conservation in industries depend on the kind of fluid involves in heat exchange process. Shell and tube heat exchanger

(STHE) is used for the liquid to liquid or gas to liquid heat transfer while compact heat exchanger (CHE) is used for gas to gas, gas to liquid or liquid to liquid heat transfer [1]. One of the important CHE is plate heat exchanger (PHE). PHEs are widely used in Petroleum, chemical processing, food & beverages, cryogenics, and pharmaceutical industries. The superior features of PHEs are its high surface area density and thermal effectiveness, resulting in reduced size, weight, and space compared to other types of heat exchanger [2]. On the other end, high hydraulic losses (i.e. pressure drop) involved in PHEs. Thus, the trade-off between thermal and hydraulic behaviour is always required to reach at optimum design of PHEs. Further, large number of design parameters is involved in the design of PHEs that should satisfy the geometric/operating constraints and heat duty requirements. As a result, metaheuristic algorithms are more suitable to obtain the optimized design of PHEs as compared to conventional optimization methods. Generally, objectives involved in the design optimization of PHE are thermodynamics (i.e. maximum effectiveness, minimum entropy generation rate, minimum pressure drop, etc.) and economics (i.e. minimum cost, minimum weight, etc.).

For PHEs, a very large number of low arrangements are possible, depending on the perforations of the plates and gasket design. The fluid entering the plate pack at its top or bottom is split into N parallel flow channels and is collected the flow in the duct on the opposite side. This group of channels in which the flow is in the same direction is denominated “pass”. Both hot and cold sides of the PHE consist of a series of passes, which usually have a constant number of channels per pass. The flow arrangement of a PHE is often presented as:

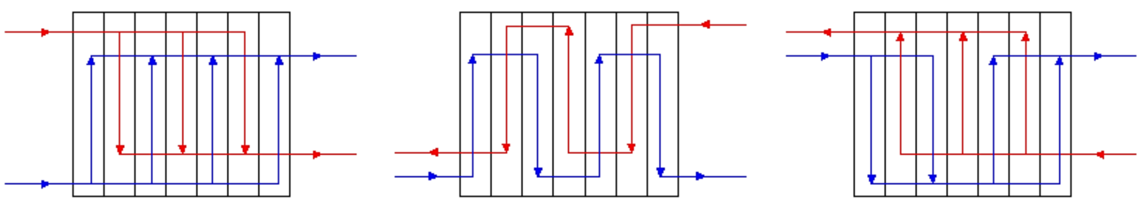
$$P_{hot} \cdot N_{hot} / P_{cold} \cdot N_{cold}$$

where P is the number of passes, N is the number of channels.

The number of channels in the hot or cold sides of the PHE is given by P*N, the total number of the channels in the PHE is thus defined as:

$$N_c = P_{hot} \cdot N_{hot} + P_{cold} \cdot N_{cold}$$

And the number of thermal plates is given by N_c-1. Figure 1 shows a few examples of possible configuration for a PHE with seven channels. The configuration consists of the flow arrangement and the location of the inlet and outlet connections of hot and cold fluids.



(a) (b) (c)

Figure 1. Some possible configurations for a PHE with seven flow channels (six thermal plates): (a) — 1*4/1*3 (looped Z); (b) — 4*1/3*1 (series); (c) — 2*2/1*3 (complex)

Overview of information sources on the problem scientific research

Earlier, researchers had carried out different types of numerical works to optimize PHEs design with different methodologies. Hajabdollahi et al. [3] obtained optimized geometric parameters of gasket plate heat exchanger for maximum effectiveness and minimum total cost by adapting NSGA-II. Hajabdollahi et al. [4] presented the comparative study of gasket plate and shell and tube heat exchangers from the economic point of view by using a genetic algorithm (GA). Najafi and Najafi [5] performed a multi-objective optimization of PHE with pressure drop and heat transfer coefficient of a heat exchanger as objective functions. The authors used NSGA-II as an optimization tool. Lee and Lee [6] carried out a thermodynamic optimization of PHE. They considered two conflicting objectives namely, Colburn factor and friction factor for optimization and used GA as an optimization tool. Further, authors also developed the correlation for Colburn factor and friction factor. Arsenyeva et al. [7] proposed mathematical model based area optimization of a multi-pass plate-and-frame heat exchanger. Gut and Pinto [8, 9] presented a mathematical model of gasket plate heat exchanger [8] and perform shape optimization [9] of that model. Further, they presented a screening method for selection of optimal configurations of plate heat exchangers. Wang and Sunden [10] used derivative-based optimization method for the economic optimization of plate heat exchanger. Durmus et al. [11] carried out an experimental investigation of plate heat exchanger having different surface geometry. They proposed heat transfer, friction factor and exergy loss correlations for plate heat exchanger based on experimental results. Zhu and Zhang [12] perform heat transfer area optimization of plate heat exchanger used for the geothermal heating application.

The thermodynamic irreversibility in a heat exchanger arises from heat transfer across a finite temperature difference among the streams. Optimizing a heat exchanger based on this concept means minimizing the amount of lost useful power. Mishra et al. [13] incorporated a genetic-algorithm-based optimization technique for a cross-flow plate and fin heat exchanger to minimize the total entropy generated within a prescribed heat duty. Cheng [14] proposed the concept of entropy resistance based on the entropy generation analysis as an alternative method for heat exchanger analysis. Fakheri [15] used the second law of thermodynamics to determine the exchanger thermal efficiency through the ratio of the actual heat transfer rate to the optimum heat transfer rate. Fakheri [16] also extended this concept to determine the efficiency of heat exchanger networks excluding the need for charts, or complicated performance statements. Apart from plate heat exchanger, efforts are put by researchers to optimize other types of heat exchangers with different objectives and methodology. For example, Patel and Rao performed optimization of shell and tube heat exchanger [17–20], plate-fin heat exchanger [20, 21], and regenerative heat exchanger [24] with different optimization algorithms. Patel and Savsani [23] presented multi-objective optimization of a plate-fin heat exchanger. Nobile et al. Performed multi-objective optimization of convective periodic and wavy channels [24–27], numerical analysis of fluid flow and heat transfer in periodic wavy channels [28], and shape optimization of a tube bundle in crossflow [29, 30].

In this study, the thermal-hydraulic characters of the plate heat exchangers on heat transfer and pressure drop, in Nu, f, E, Pe numbers base, is experimentally examined.

Experimental setup procedure

Details of the experimental set-up of PHE are shown in the schematic layout (Figure 2). Red and blue arrow marks with trace the hot and cold water respectively. Circulating water is stored in electric boiler (1), condenser (17), and cooling tower (19). Reciprocating pump (2) propels the heated water which comes from the electric boiler and heats to the desired temperature with 0.1 L/s to the plate heat exchanger where it is converted with cold water, Reciprocating pump (16) propels the cooled water which comes from the condenser and cools to the desired temperature with 0.22 L/s to the plate heat exchanger where it is converted with hot water. Maximum operating pressure and temperature of the electric boiler are 4 Ba and 80°C.

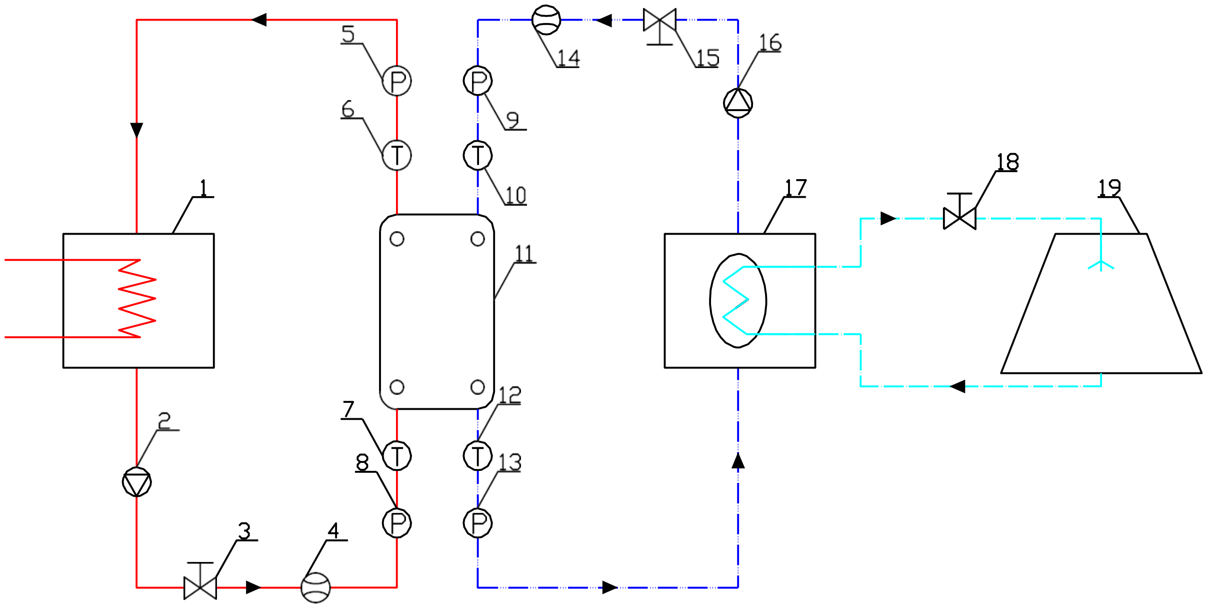


Figure 2. Schematic layout of experimental set-up: 1 — electric boiler; 2 — first pump; 3 — first vale; 4 — first flow meter; 5 — first pressure transmitter; 6 — first temperature transmitter; 7 — second temperature transmitter; 8 — second pressure transmitter; 9 — third pressure transmitter; 10 — third temperature transmitter; 11 — PHEs; 12-fourth temperature transmitter; 13 — fourth pressure transmitter; 14 — second flow meter; 15 — second valve; 16-second pump; 17 — condenser; 18-third valve; 19 — cooling tower.

Investigation of the flow distribution in PHE is carried out for 13 plates (12 channels). The turbine type rotary mechanical flow meters (4 & 14) and four PT-100 thermometers (6, 7, 10 & 12) are provided in hot and cold water flow path to measure the volume flow rates and temperatures, respectively. Thermometers are placed near to the ports of PHE in the stainless steel pipe section at the inlet and outlet of water. Four pressure transmitters (5, 8, 9 & 13) are placed near the thermometers to measure the fluid pressure. Flow rates of the working fluid being pumped by pumps (2 & 16), is controlled by the pump's speed. Mechanical flow meters (turbine type flow meter), pressure transmitters and thermometers are connected to a Programming Logic Controller (PLC), which is further connected to a human interface unit (HMI), measure and control the flow rates of hot and cold fluid streams. They also measure the pressure and temperature at the inlet and outlet of both the fluid streams.

Figure 3. Boiler.

A pictorial view of the experimental test section as shown in Figure 4. It consists of PHE, Turbine type mechanical flow meters (10 & 11) and M5100 pressure transmit ters (2, 3, 4 & 5) are provided in hot and cold fluid stream as already discussed. Thermometers (6, 7, 8 & 9) are provided near the inlet and outlet port of cold and hot fluid streams. The working fluid is pumped by pump.

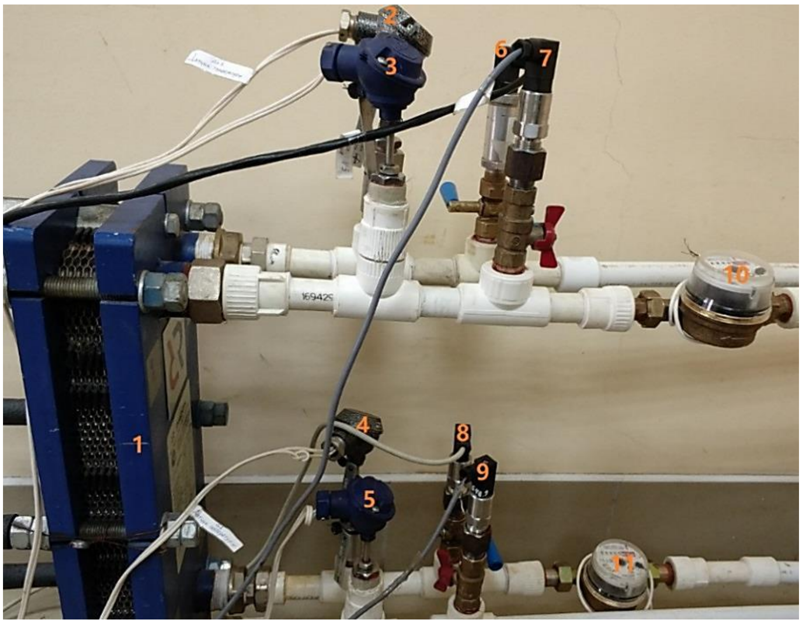
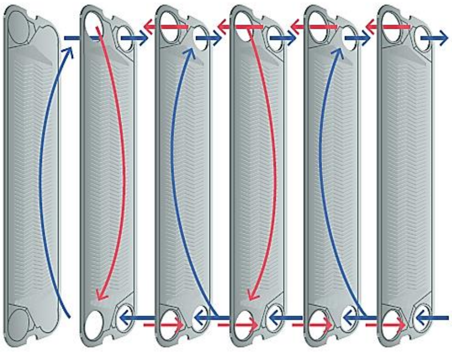


Figure 4. Experimental test section—plate heat exchanger; 2, 3, 4, 5 — M5100 pressure transmitters; 6, 7, 8, 9 — thermometers; 10, 11 — mechanical flow meters

Figure 5 presents the flow arrangement of PHE: (a) geometrical and flow path details and (b) shows the details of one plate. Geometrical parameters of the same are listed in Table 1.



(a)

(b)

Figure 5. Flow arrangement for PHE and details for plate heat exchanger: (a) — side view of plates with the flows arrangement of cold and hot water of PHE; (b) — front view of the plate

Table 1.

GEOMETRICAL CHARACTERISTICS OF CHEVRON PLATE

Geometrical characteristics of chevron plate		
Number	*Particulars*	*Dimensions*
1	Port diameter, d_p	0.032m
2	Port to port length, L_{ch}	0.381m
3	Port to port width, Lw	0.07m
4	Corrugation picth p	0.011m
5	Amplitude of corrugation, b	0.0028m
6	Thichness of plate, t	0.0005m
7	Chevron angle, β	60°
8	Heat transfer area (plate)	0.04m²
9	Plate material	AISI 316 (Stainless steel)
10	Gasket material	Nitrile rubber

In isothermal condition, cold water is supplied to all the channels of PHE, thus there is no heat transfer involved in the PHE. This analysis is carried out to study hydraulic performance at a constant temperature. The experiments have been carried out at 12°C for the isothermal condition. In non-isothermal condition, cold and hot water are supplied in alternative channels of the PHE. Cold water is supplied at 12°C and hot water is maintained in the range of 70-71°C.

The details of measuring instruments used in the experiment are shown in Table 2 below.

Table 2.

THE MODEL, RANGE, AND ACCURACY OF THE MEASURING INSTRUMENTS

Number	*Instrument*	*Measure*	*Model*	*Range*
1	Mechanical flow meters	Flow rate	Pt 500	1000-30000 LPH
2	Thermocouple	Temperature	PT-100, R.T.D	0 to 180 ℃
3	Pressure transmitters	Inlet and outlet pressure of PHEs	M5100	0-2.5 MPa

Data reduction

The experimental data have been obtained under steady state conditions and the operating flow rates were taken to have a range of Reynolds number from 220 to 2400 for PHE. The mean pressure drop data obtained using pressure transmitters was used in the following equation to evaluate Fanning friction factor (f)

$$f = \frac{\Delta P_{ch} D_{eq}}{2L_{\mathrm{ch}} \rho V^2}$$

Where ΔP is pressure drop, Pa; D_{eq} is hydraulic diameter, m; L_{ch} is port to port length, m; ρ is the density of the water, kg/m^3 ; V is the velocity of the water, m/s.

The measured pressure drop of PHE includes the frictional pressure drop in the channels and the pressure drop in the ports.

$$\Delta P = \Delta P_{ch} + \Delta P_p$$

Where ΔP_p is pressure drop in port?

The channel pressure drop is dined by using the Darcy friction factor model as below:

$$\Delta P_{ch} = 4f\left(\frac{L_p}{D_{eq}} \rho \frac{V^2}{2}\right)$$

where f is the channel friction factor that is determined by correlation obtained from Focke et al. [31]; L_p is the port to port length, m.

To calculate the pressure drop in the ports Shah and Focke [32] correlation is used as follows:

$$\Delta P_p = 1.5 \frac{\rho u^2}{2}$$

Where the u is velocity in port, m/s.

The behavior of hydraulic resistance with Reynolds number is shown in Figure 9. The correlation for the Fanning friction factor in terms of Reynolds number is developed in fitting formula (1). The Reynolds number for plate heat exchanger is defined on the basis of hydraulic diameter D_{eq} , as:

$$\mathrm{Re} = \frac{V D_{eq}}{\upsilon}$$

where υ is Kinematic viscosity, m^2/s.

Reynolds numbers were randomly selected within this range. The characteristic length of the channel was the equivalent diameter, which is defined as follows:

$$D_{eq} = \frac{4bL_w}{2(b + L_w \phi)} \approx \frac{2b}{\phi}$$

Where the ϕ is enlargement factor.

The inlet port velocity is evaluated as

$$V = \frac{V_{h.i}}{A_p * n}$$

Where $V_{h.i}$ inlet volume flow is rate, m^3/s; A_p is the inlet sectional area, m^2; n is number of channels per pass.

With respect to the average heat transfer coefficient (h), it was calculated by:

$$h_i = \frac{Q_i}{S_i \Delta T_m}$$

where Q_i is the heat transfer rate (use $Q = \frac{Q_h + Q_c}{2}$ to calculate), W; S_i is the heating wall surface area, m^2;

ΔT_m is the logarithmic mean temperature difference which is given by:

$$\Delta T_m = \frac{(T_{h.in} - T_{c.out}) - (T_{h.out} - T_{c.in})}{Ln\left[\frac{T_{h.in} - T_{c.out}}{T_{h.out} - T_{c.in}}\right]}$$

where $T_{c.in}$ is cold inlet flow temperature, K; $T_{c.out}$ is cold outlet flow temperature, K; $T_{h.in}$ is hot inlet flow temperature, K; $T_{h.out}$ is hot outlet flow temperature, K.

In the heat transfer at a surface within a fluid, the Nusselt number (Nu) is the ratio of the convective to the conductive heat transfer across normal to the boundary and it is given by:

$$Nu = \frac{hD_{eq}}{\lambda}$$

where h is the convective heat transfer coefficient, w/(m^2*k); λ is the thermal conductivity, w/(m*k).

The Prandtl number ($\Pr$) is the ratio of the momentum and the thermal diffusivities, and it describes the static properties of the fluid substance. The generalized Prandtl number is defined by:

$$\Pr = \frac{\mu C_p}{\lambda}$$

where C_p is the specific heat, J/(kg*k); μ is Newtonian viscosity, Pa*s.

Also, the Peclet number is defined as:

$$Pe = \mathrm{Re}\Pr$$

Theoretical analysis for exergy loss

Exergy is called the amount of maximum work obtained theoretically at the end of a reversible process in which equilibrium with environment should be obtained. According to this definition, exergy balance in a steady-state open system can be written as follows:

$$E = T_e S_{irr} = T_e\left[\dot{m}_c\left(S_{c,out} - S_{c,in}\right) + \dot{m}_h\left(S_{h.out} - S_{h,in}\right)\right]$$

where T_e is environment temperature, k; S_{irr} is entropy production, J/mol*k; $\dot{m}$ is water mass flow, kg/s; S is entropy, J/mol*k.

$$dS = \int_{v=const}\frac{dQ}{T} = \int_{v=const}\frac{du + pdv}{T} = \int_{v=const}\frac{CdT}{T}$$

for liquids (v = const.);

Entropy production of adiabatic heat exchanger:

$$S_{irr} = C_h In\frac{T_{h,out}}{T_{h,in}} + C_c In\frac{T_{c,out}}{T_{c,in}}$$

where C is heat capacity rate, c=mc$_p$, w/k.

Exergy loss;

$$E = T_e \left[C_h In \frac{T_{h,out}}{T_{h,in}} + C_c In \frac{T_{c,out}}{T_{c,in}} \right]$$

We can directly be determined. For liquids E contains also the exergy loss caused by pressure drop.

Results and discussion

Figure 6 shows the variation of Nusselt number in the hot side and cold side with respect to Reynolds number for PHE. Nusselt numbers in the cold side were found to be less than that in the hot side (Figure 6). Because velocity in hot side bigger than cold side, and it also shows change same velocity range in different temperature has different impact on heat transfer. Then the gradient in cold side is large than hot side. The empirical correction is showed.

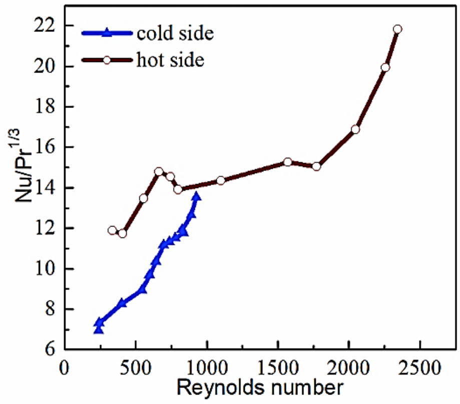


Figure 6. Nusselt number as a function of Reynolds number

Hot side:

$$Nu = 2.87\,\mathrm{Re}^{0.2423}\mathrm{Pr}^{0.3} \qquad 340 \le \mathrm{Re} \le 2400$$

Cold side:

$$Nu = 1.57\,\mathrm{Re}^{0.3071}\mathrm{Pr}^{0.4} \qquad 240 \le \mathrm{Re} \le 930$$

Figure 7 show the variation of the exergy loss with different inlet water volume flow rates. The experimental results show that heat transfer has great effect on exergy loss. It is clear seen that exergy loss of plate heat exchanger is increase rapidly with flow rate. And exergy loss in hot side big than cold side when flow rate from 0.008-0.2 L/S, cold side more large than hot side after 0.2 L/S.

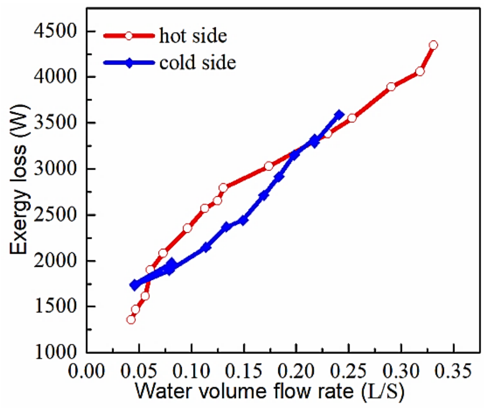


Figure 7. Variations of exergy loss with water volume flow rate

Figure 8 shows the pressure drop and convective heat transfer coefficient versus Reynolds number at different velocity in hot side. It can be seen, the pressure drop rises with the increase of Reynolds number. This augmentation can be related to enhancement of viscosity and density of the water and resistance of the channel. By the way, based on these results the convective heat transfer coefficient increases volatility with enhancement of Reynolds number. It maybe depends on increasing of turbulence intensity at higher Reynolds number and decrement in fluid thermal boundary layer thickness due to the reduction of fluid viscosity near the wall, and the thermal boundary layer is unstable in transition. Also heat exchange comes into full development stage and thermal boundary layer remains unchanged after Reynolds number more than 1800.

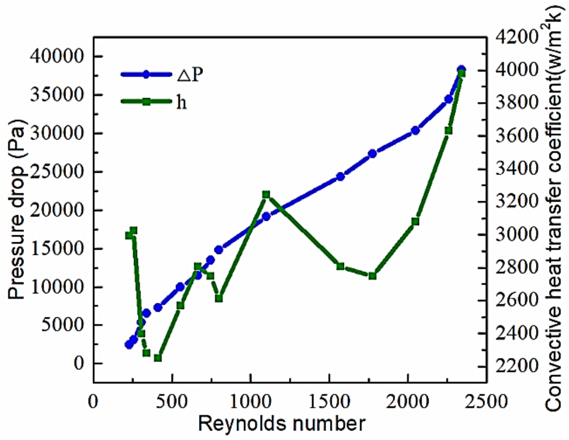


Figure 8. Pressure drop and convective heat transfer coefficient changed with Reynolds number

Figure 9 shows the behavior of the Fanning friction factor with Reynolds number in hot side. For the Reynolds number range investigated in this work, the following empirical correlation of the Fanning friction factor as a function of the generalized Reynolds number is proposed.

$$f = 2463.9\,\mathrm{Re}^{-0.8147} \quad 220 \leq \mathrm{Re} \leq 2400 \tag{1}$$

It is observed that Fanning friction factor decreases with an increase in the Reynolds number. This is due to a tremendous increase in turbulence at higher Reynolds number within plates. At higher Reynolds number, fluid molecules get lesser time to interact with plate surface, and hence lower friction between the plates and fluid particles. The fluctuations in the Fanning friction factor at higher Reynolds number are due to the experimental error like the vibration in the system when the motor speed goes above 2500 RPM.

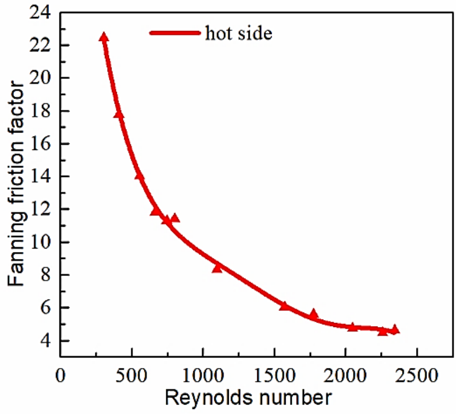


Figure 9. Variations of Fanning friction factor with Reynolds number

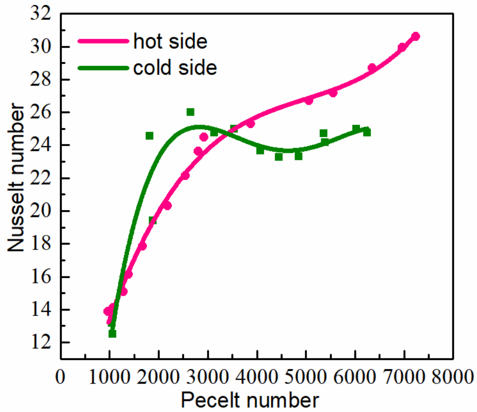


Figure 10. Behavior of the Nusselt number as a function on the Peclet number in PHE.

The variation of the Nusselt number as a function of the Peclet number (Pe = RePr) for all the case studies with plate heat exchanger is presented in Figure 10, in which it is possible to observe that the increase of the Peclet number results in an increase of the Nusselt number as well when Peclet number small than 2000 in cold side. However, Nusselt number sustainable increase with Peclet number. Nusselt number in cold side bigger than hot side at Peclet number's range from 1000 to 3000, and hot side more greater than cold side after Peclet number is greater than 3000. The increase of the Nusselt number indicates an enhancement in the heat transfer coefficient due to the convection increases. That is mean we get reference idea (improving velocity in hot side) how to improve heat transfer efficiency in the particular case using plate heat exchanger.

Conclusion

In this study, in order to research heat transfer in plate heat exchanger. Nusselt number, the efficiency, pressure drop and exergy loss is discussed comparatively. The conclusions can be drawn from the experimental study, and show the efficiency of the heat exchanger increases with increasing the fluid' volume flow rates and pressure drop due to an enhanced heat transfer. In the experimental results obtained the empirical corrections about Reynolds number, Prandtl number and Nusselt number, Fanning friction factor decreases with volume flow rate, thus convective heat transfer coefficient and exergy loss increase with volume flow rate. Accordingly, pressure drop increases too. Pressure drop greatly increases the capital costs. Because of this, a thermodynamic optimization should be made between heat transfer and pressure drop. Pressure drop, however, has not an importance for heat exchanger. Because, increasing of the heat exchanger efficiency causes smaller dimensions of the heat exchanger and decreasing of the production costs.

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